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ABSTRACT

This report presents a preliminary evaluation of a Brayton cycle turboalternator using gas-lubricated bearings and operating on argon at a turbine inlet temperature of 1225° F (936 K). The turboalternator was operated successfully through steady-state and transient modes for 1165 total hours, during which time it was started and stopped 24 times. The alternator output was 8.8 kilowatts electric at turbine design conditions; the maximum output obtained was 33.2 kilowatts electric. The temperatures in the turboalternator did not exceed the predicted values when the turboalternator was operating at design conditions. The bearings always operated within the design clearance limits during both steady-state and transient testing. Largest clearance changes were experienced with the thrust bearings during full-load electrical transient tests.

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SUMMARY

This report presents a preliminary evaluation of a Brayton cycle turboalternator using gas-lubricated bearings and operating on argon at a turbine inlet temperature of 1225° F (936 K).

The turboalternator was operated successfully through steady-state and transient modes for 1165 total hours, during which time it was started and stopped 24 times. The alternator output was 8.8 kilowatts electric at turbine design conditions; the maximum output obtained was 33.2 kilowatts electric. The temperatures in the turboalternator did not exceed the predicted values when the turboalternator was operating at design conditions.

The bearings always operated within the design clearance limits during both steady-state and transient testing. Largest clearance changes were experienced with the thrust bearings during full-load electrical transient tests.

INTRODUCTION

The NASA Lewis Research Center is studying electrical power conversion systems for space use. One system operating on the Brayton thermodynamic cycle and using argon as the working fluid is described in reference 1. This version of the cycle uses a turbine-driven compressor to circulate the argon and a turbine-driven alternator (turboalternator) to generate electrical power. The turboalternator shaft is supported by gas-lubricated bearings because this eliminates possible contamination of the system gas with oil and the need for a complex oil lubrication system. These are important criteria for a space power system that may be required to run unattended for periods of a year or more.

The turboalternator was designed and built under contract (refs. 2 and 3). During the design of the turboalternator, several experimental machines were built to evaluate separately the design of the turbine (ref. 4), alternator (ref. 5), and gas bearings (refs. 2 and 3). References 2 and 3 also discuss testing done on the alternator - gas-bearing combination.

Testing of the turboalternator at the Lewis Research Center and a preliminary evaluation of its performance are reported herein. The primary emphasis is placed on the performance of the gas bearings.

The major problems were to be able (1) to start and stop turboalternator shaft rotation without bearing damage, (2) to maintain proper bearing clearance during the startup thermal transient from room temperature to design turbine inlet temperature of 1225° F (936 K), and (3) to have good shaft stability under the loads imposed by shaft dynamics and the alternator magnetic field. Stability is important because gas bearings have very low damping capability. Previous tests described in references 2 and 3 showed good shaft stability and bearing clearance control at alternator outputs up to 12 kilowatts electric. But all these tests were made with turbine inlet temperatures not exceeding 200° F (366 K). Therefore, the primary objective at Lewis was to operate the turboalternator at the design turbine inlet temperature of 1225° F (936 K). Other objectives included the evaluation of overall performance and gas-bearing performance for a period of 1000 hours.

TEST FACILITY DESCRIPTION

The test facility was built to operate the turboalternator at its design turbine inlet conditions. The facility is a closed loop using a motor-driven compressor to circulate the working fluid argon. A flow schematic is shown in figure 1.

The argon supplied to the turbine was heated by the electric heater. Inlet pressure was controlled by regulating the flow through the turbine bypass. Turbine weight flow was measured by the venturi upstream of the turboalternator. Pressure ratio was controlled by the valve at the turbine discharge. This control valve was also used to control turbine speed for some of the tests.

The oil coolant loop supplied cooling oil to the alternator stator and to heat exchangers at each of the bearings. Oil flow rate to the alternator and bearing heat exchangers could be controlled individually. Coolant temperature could also be regulated. A small amount of argon flowing through the turboalternator was also required. This argon was supplied from external high-pressure argon bottles with regulators controlling pressure.

A schematic of the electrical system used to absorb the turboalternator output power

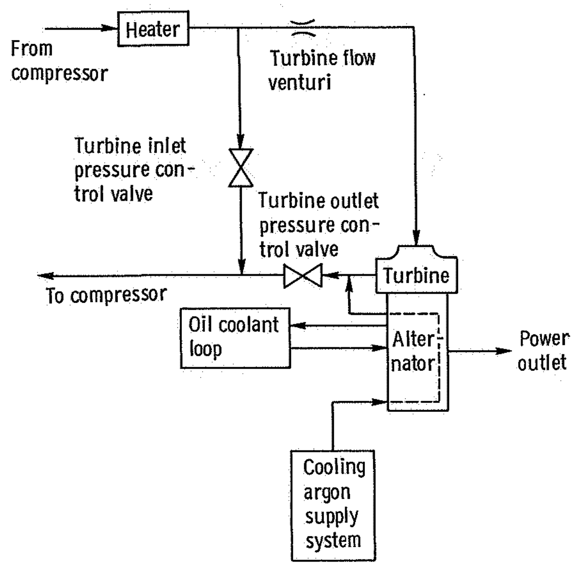


Figure 1. - Schematic of Brayton cycle turboalternator test facility.

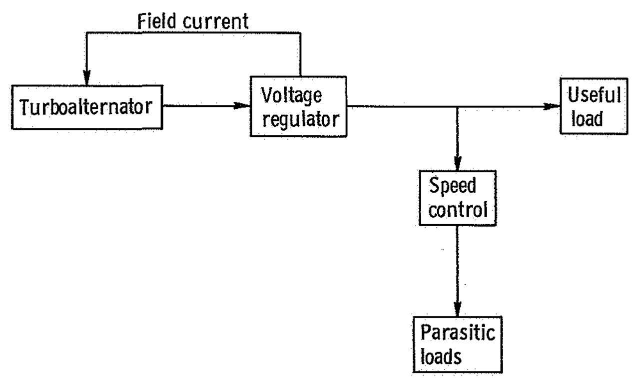


Figure 2. - Schematic diagram of turboalternator electrical system.

is shown in figure 2. The total alternator output passes through the voltage regulator and is absorbed by the speed control parasitic load bank and by the load bank simulating a useful power systems load.

The voltage regulator described in reference 6 supplies the required current to the alternator field to maintain alternator output voltage at the rated line voltage over the normal alternator output power range. The regulator is made up of solid-state electronic components.

The electric speed control system described in references 7 and 8 maintains alternator output frequency by controlling alternator load. The load is dissipated by the parasitic load bank. The speed control is also made up of solid-state electronic components.

The useful load bank is capable of applying balanced and unbalanced loads at lagging power factor of 1.0 to 0.6 and step-load transients.

INSTRUMENTATION DESCRIPTION

The instrumentation necessary to evaluate the performance of the turboalternator consisted of thermocouples, pressure transducers, flowmeters, distance-measurement probes, and electrical power instruments.

Chromel-Alumel thermocouples were used to measure turboalternator internal temperatures and turbine inlet and outlet gas temperatures. The inlet and outlet gas temperatures were measured with bare spike stream thermocouples mounted on rakes. The oil coolant temperatures were measured with iron-constantan thermocouples immersed in the oil stream. The temperatures could be read to within 1 percent.

Pressures were measured with strain gage transducers. Static and total pressures were measured at the turbine inlet and outlet. Other pressures were measured in the turboalternator case. The transducers were calibrated with a Bourdon tube gage having an absolute error of ±0.030 psi ($\pm 0.207\times10^3$ N/m^2). The pressure transducers have a precision of ±0.015 psi ($\pm 0.103\times10^3$ N/m^2).

The turbine weight flow was measured with a calibrated venturi. The venturi pressures were measured with the strain gage transducers. Coolant oil flow was measured with turbine flowmeters. Coolant gas flows were measured with rotameters. The rotameters and the turbine flowmeters were accurate within 3 percent.

Bearing static and dynamic motions and shaft motions in the turboalternator were measured with capacitance probes. A total of 21 capacitance probes were mounted on the turboalternator. They measure journal motion relative to the housing, thrust bearing film thickness, film thickness between each bearing pad and the shaft, and the dynamic motion of the bearing pads relative to the turboalternator housing.

The capacitance probe measures the electrical capacitance of the gas gap between the probe tip and the opposite surface. The output of the probe unit is a voltage that is directly proportional to the gap width. The absolute error is 0.2 mil (5.1×10^{-4} cm); the precision is 0.025 mil (0.64×10^{-4} cm). All the outputs of the capacitance probe units are recorded on an FM magnetic tape recorder. The outputs of the bearing clearance probes were also sent to oscilloscopes for continuous monitoring.

The wattmeters, voltmeters, and ammeters used to measure alternator output were the wide frequency range, true rms electronic type. Their accuracy was within 1 percent or better. The ac voltage and current signals were converted to dc by solid-state thermocouple converters. The wattmeters were electronic electrodynamometer converters supplying dc signals.

Turboalternator shaft speed was measured with three magnetic speed pickups. The pickups received the signal from six notches in the thrust bearing runner.

TURBOALTERNATOR DESCRIPTION

The turboalternator consists of a four-pole homopolar alternator driven by a two-stage axial-flow turbine. The turbine and alternator are mounted on a single shaft that is supported by two gas-lubricated journal bearings and a gas-lubricated thrust bearing. The turboalternator is shown in figure 3. The turboalternator design criteria are listed in table I. A cross-sectional view is shown in figure 4.

The alternator is a four-pole radial-gap solid-rotor brushless homopolar inductor. The field coil is located in the stator between the rotor poles. The alternator was designed to minimize magnetic unbalance forces and maintain a high efficiency. The high efficiency called for a conservative electromagnetic and thermal design. A relatively large rotor gap of 0.40 inch (0.1 cm) was used to reduce magnetic unbalance forces. The stator is liquid cooled.

Each of the journal bearings consists of four tilting pads each pivoting on a ball-and-socket joint. The ball-and-sockets have nonconforming radii. The bearing pads and joints are mounted to the bearing-support ring with flexible mounts (flexures). The flexures allow for a limited amount of radial thermal growth and have provisions for mechan-

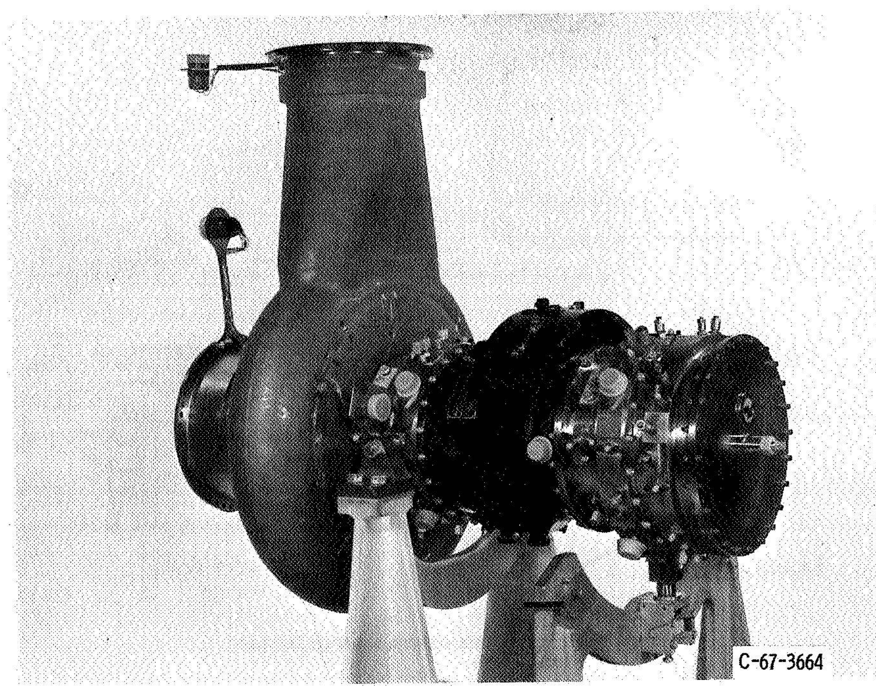


Figure 3. - Turboalternator.

TABLE I. - TURBOALTERNATOR DESIGN CONDITIONS

Turbine:	
Working fluid	Argon
Weight flow, lb/sec (kg/sec)	0.611 (0.277)
Inlet total temperature, ^{o}F (K)	1225 (936)
Inlet total pressure, psia (N/cm^2)	8.45 (5.83×10^4)
Pressure ratio, total to static	1.26
Alternator:	
Design power, kVA	15 at 0.8 power factor
Number of phases	3
Voltage, V	120/208
Frequency, Hz	400
Stator hot-spot temperature, ^{o}F (K)	356 (453)
Turboalternator:	
Shaft speed, rpm	12 000
Shaft overspeed, rpm	14 400
Coolant flow, lb/min (kg/min)	18.9 (00.0)
Coolant temperature, ^{o}F (K)	200 (367)

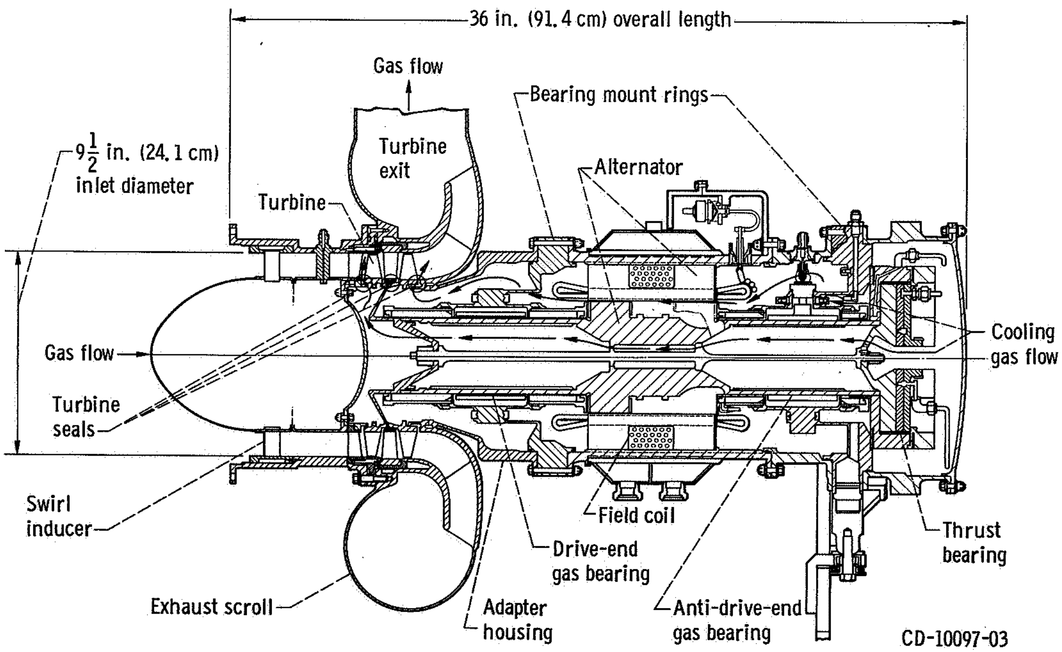


Figure 4. - Cross-sectional view of turboalternator.

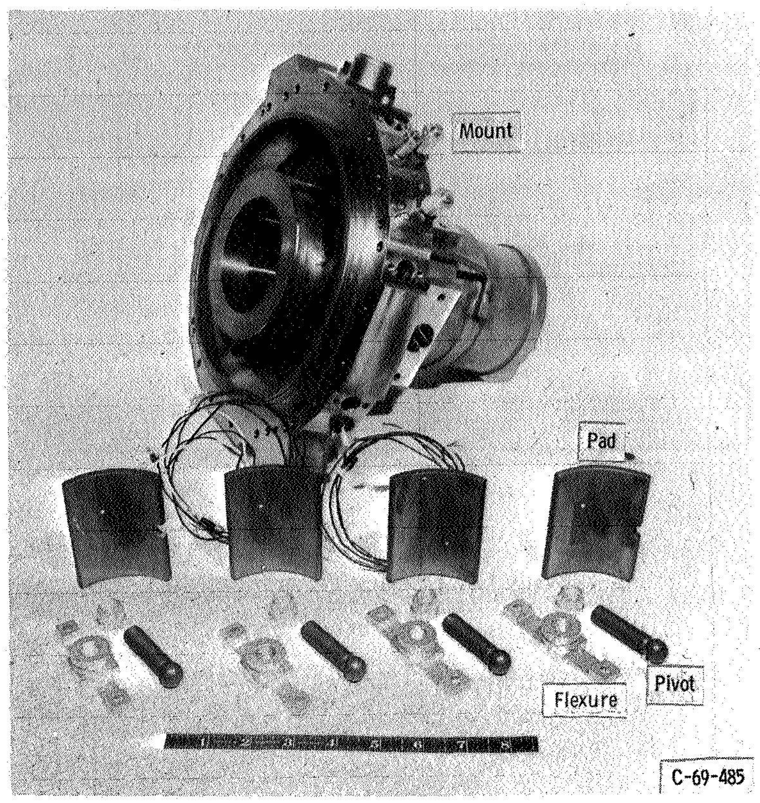


Figure 5. - Parts layout for journal bearing assembly.

ically adjusting bearing clearance. The bearing parts are shown in figure 5. Two adjacent pads of each bearing have provisions for external pressurization (hydrostatic jacking gas). These are used only for starting and stopping the turboalternator when it is in a position other than vertical.

A schematic of the journal bearing design is shown in figure 6. Journal bearing heat is removed by heat exchanger sleeves around the shaft on each side of the bearings. The heat is conducted from the shaft to the heat exchangers through a 0.005-inch (1.27×10^{-2}-cm) radial gap. The inside of the shaft under the journal bearing area is plated with a 0.125-inch (0.318-cm) thick layer of copper. The plating is used to maintain uniform temperatures along the shaft under the bearing pads.

Each of the journal-bearing-support rings also has liquid heat exchangers. By variation of the fluid temperature in these heat exchangers, the support ring size can be changed to control the bearing clearance.

A schematic view of the thrust bearing is shown in figure 7. Figure 8 shows the main thrust stator. The thrust bearing is designed to take load in both axial directions. The normal thrust load is toward the rear of the turboalternator. The thrust load is 87 pounds (390 N) when the turboalternator is in the vertical position; 57 pounds (25 kg) is shaft weight; 30 pounds (130 N) is aerodynamic forces. The main thrust bearing is

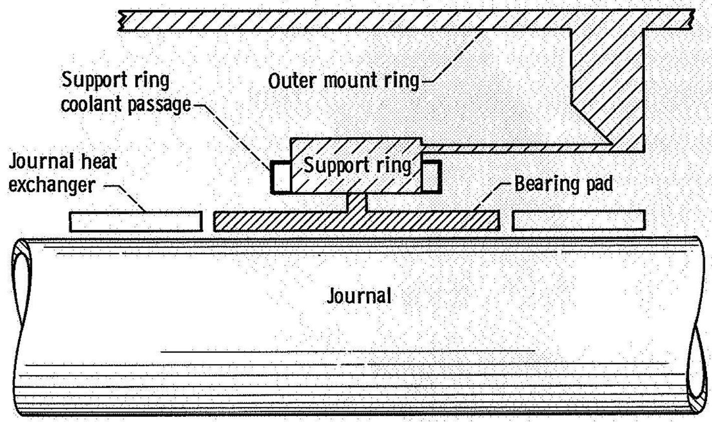


Figure 6. - Journal bearing schematic.

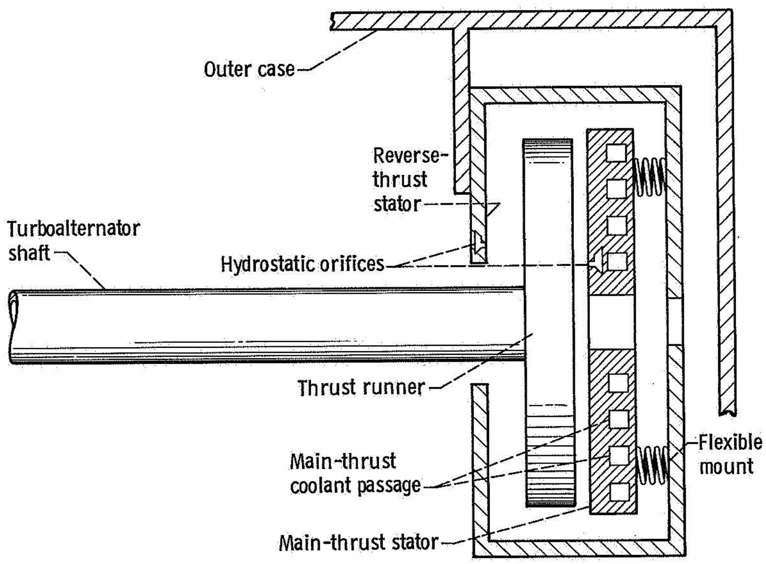


Figure 7. - Thrust bearing schematic.

designed for a maximum 250-pound (1100-N) load to provide for transient loads. The reverse thrust bearing is designed for 100 pounds maximum (440 N). The main thrust bearing is designed to be self-acting (hydrodynamic); however, both main and reverse thrust stators have hydrostatic capabilities. The main thrust bearing stator is flexibly mounted to permit dynamic alinement of the stator with the thrust runner. Liquid coolant flows through the main thrust stator to cool it.

A small amount of argon obtained from the cooling supply source of figure 1 is fed through the turboalternator from the thrust bearing end of the machine. The gas flows through and around the shaft toward the turbine end (fig. 4). The gas then enters the main gas flow through the turbine seals. The primary purpose of the gas is to prevent

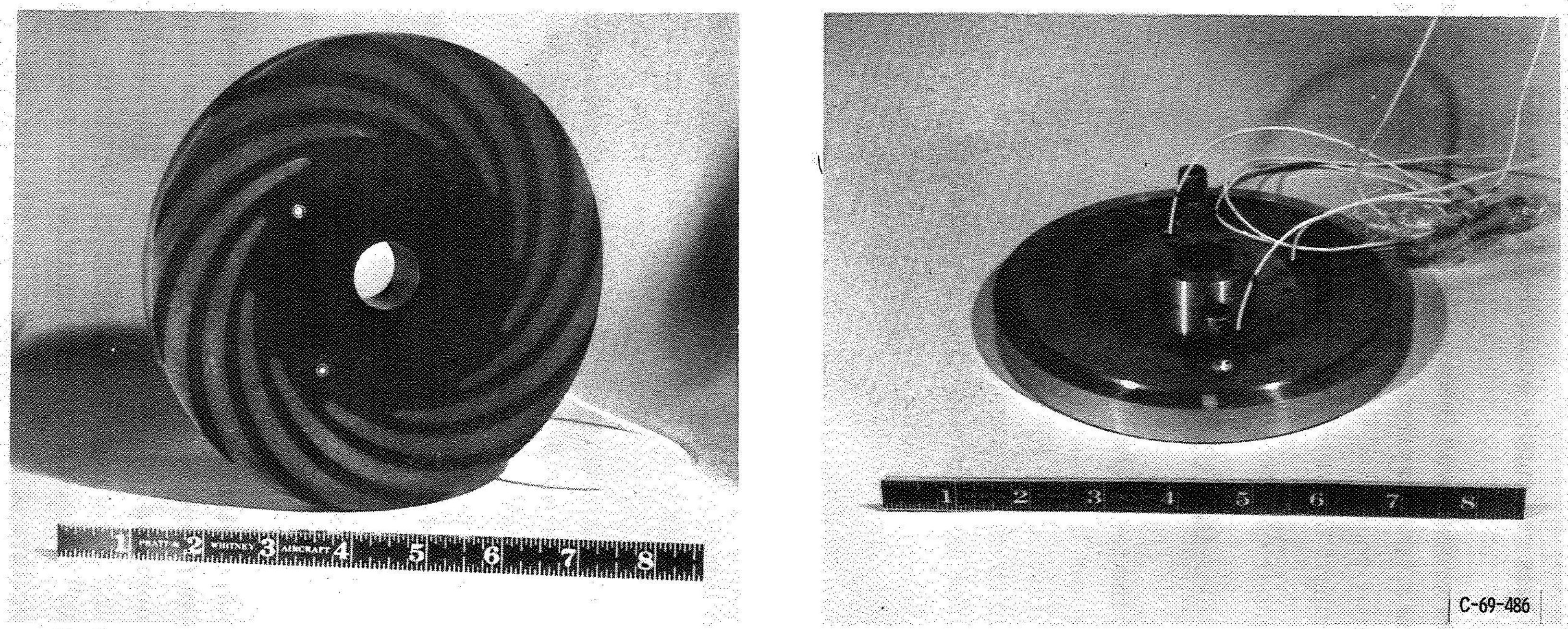


Figure 8. - Main-thrust stator.

backflow of hot turbine gas into the bearing area although it does assist in heat removal. In a space power system, this gas supply would be bled from the discharge of the argon-circulating compressor.

A special turbine inlet section was installed on the turboalternator to simulate the flow conditions expected in a power system having the turbine discharge of the argon-circulating turbocompressor connected directly to the turboalternator turbine inlet. The vanes shown in figure 4 impart the swirl expected from the turbine exhaust.

DISCUSSION OF RESULTS

The first objective of the testing was to operate the turboalternator at design conditions. This objective was met by running the turboalternator without interruption for an arbitrarily selected time duration of 100 hours. After the 100-hour test, design and off-design conditions were run to evaluate further the turboalternator and the gas bearings. The total running time on the machine was 1165 hours during which it was started and stopped 24 times.

Turbine and Alternator Performance

The effects of turbine inlet-to-outlet pressure ratio on turbine weight flow is shown in figure 9. The weight flow data were corrected for variation in inlet pressure and inlet temperature.

At the turbine-design values of inlet temperature, inlet pressure, and overall pressure ratio, the alternator output was 8.8 kilowatts electric. The design goal was 9 kilowatts electric. The results of separately testing the turbine and alternator (refs. 4 and 5) indicated the turboalternator power to be 8.7 kilowatts electric.

The alternator thermal design is conservative, producing an alternator that could run well over the rated power output of 12 kilowatts electric without exceeding rated hot-spot temperature of 180° C. At the end of the testing period the turboalternator power was therefore raised to see what power could be attained within this hot-spot temperature limit.

A plot of alternator output starting at 15 kilovolt-amperes versus the highest alternator temperature is shown in figure 10. The hottest temperature was an end-turn of the stator winding. At maximum power, it was only 186° C. The maximum power reached was 33.2 kilowatts electric. This value was limited by the facility. The maximum specification temperature of 180° C represents an insulation life of over 10 years. The output power at a hot-spot temperature of 180° C is 32 kilowatts electric.

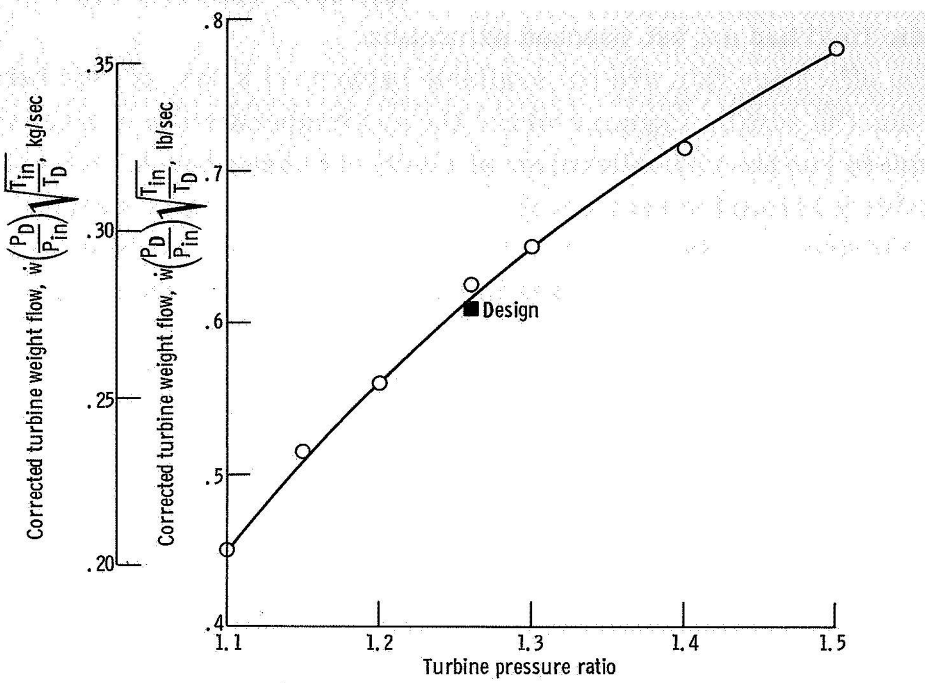


Figure 9. - Effect of turbine pressure ratio on turbine weight flow corrected to design inlet pressure and temperature for 12 000 rpm.

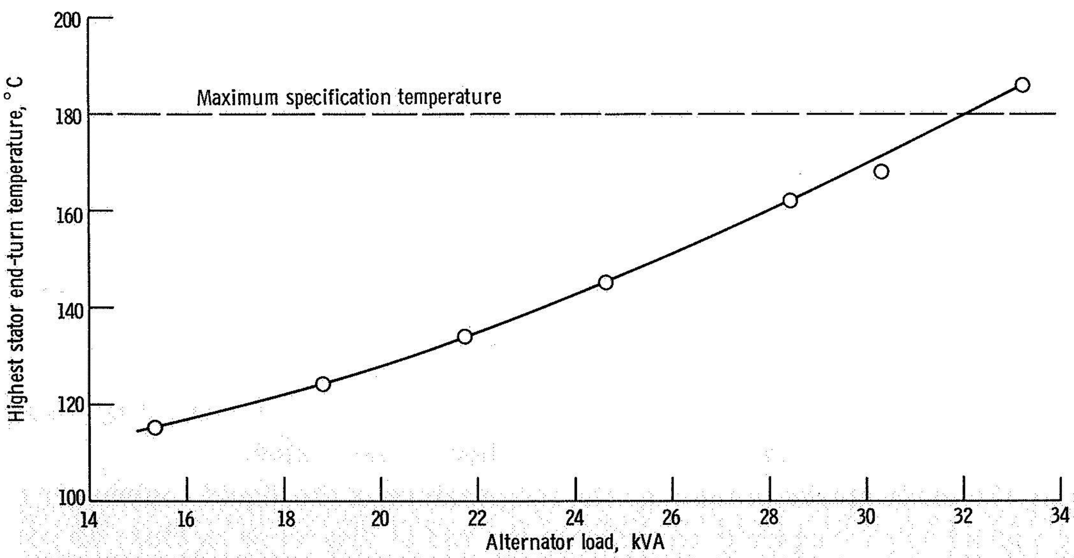


Figure 10. - Effect of alternator power on highest measured internal alternator temperature. Power factor range, 1.0 to 0.95.

The voltage regulator was not able to supply sufficient current to the alternator field to maintain proper magnetic flux at powers over 20.7 kilowatts electric, so a separate power supply was used to excite the field. At the 33.2-kilowatts-electric power level, the magnetic field had not yet reached saturation.

Turbine efficiency data are not available because of a loss of heat between the turbine discharge and the scroll discharge where the gas temperature was measured. During the first attempt to run the turboalternator at elevated temperatures, excessive heat was being transferred from the rear scroll wall to instrumentation connectors on the adapter housing of the drive-end bearing, causing the connectors to overheat. An external ring directing cooling air to the region was therefore installed in order to prevent overheating of the connectors.

Figure 11 is a cutaway view of the turboalternator showing the measured temperature distribution at design operation conditions. The values predicted in design are shown in parentheses. The arrow in figure 11 indicates the area on which the cooling air was directed. As expected, the temperatures in the area affected by the cooling air are lower than predicted. The alternator end-turn temperatures were not affected by the cooling. Since the instrumentation was vital to these tests, no testing was done at design turbine inlet temperature without using the air cooling. Thus, no conclusion can be drawn about the thermal design in the scroll and adapter housing areas because of the use of the cooling air.

Also shown in figure 11 are the bearing and alternator cavity pressures obtained with a gas flow of 0.0021 pound per second (9.5×10^{-4} kg/sec) through the turboalternator. The flow rate of 0.0021 pound per second corresponds to 0.33 percent of the turbine flow.

Bearing Performance

The turboalternator was operated with the shaft in the vertical position eliminating the effect of gravity on the journal bearings. Hydrostatic jacking gas was not used on the journal bearings throughout the 1165 hours of operation.

The operating radial clearance of the journal bearings is 0.5 to 1.5 mil (1.3×10^{-3} to 3.8×10^{-3} cm). Variation of the clearance within these limits was caused by changes in temperature of the bearing mounting and the turboalternator shaft.

During the early stages of testing, the temperature of the liquid coolant for the support ring was adjusted frequently to maintain the radial clearance as close as possible to 1 mil (2.5×10^{-3} cm). After gaining operating experience with the turboalternator, the clearance was allowed to vary more but still within the design limits. During the running of the individual bearing tests, the coolant temperatures and flows were kept as constant as possible. The clearance remained within the design limits for all the bearing tests.

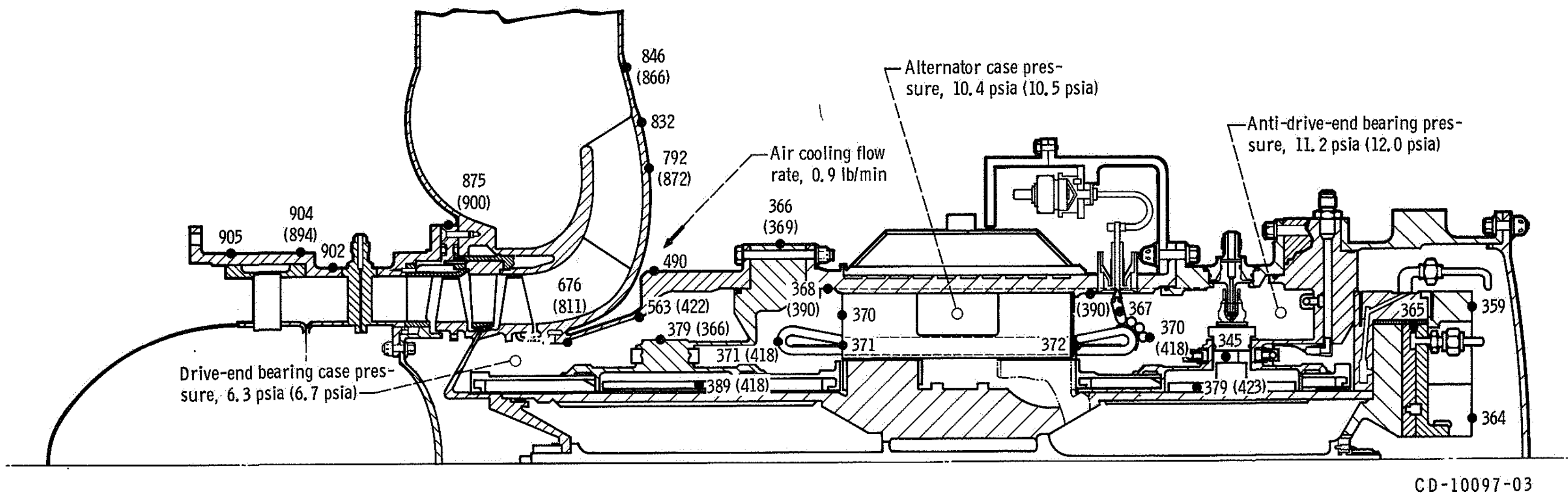


Figure 11. - Turboalternator overall thermal map and pressure distribution (predicted values in parentheses). Speed, 12 000 rpm; turbine inlet temperature, 1225° F (936 K); turbine inlet pressure, 8.45 psia (5.83×10^{4} N/cm^2); turbine pressure ratio, 1.26; alternator power, 9 kilowatts electric; power factor, 1; turbine weight flow, 0.63 pound per square second (0.286 kg/sec); turbine cooling gas weight flow, 0.0021 pound per second (9.5×10^{-4} kg/sec).

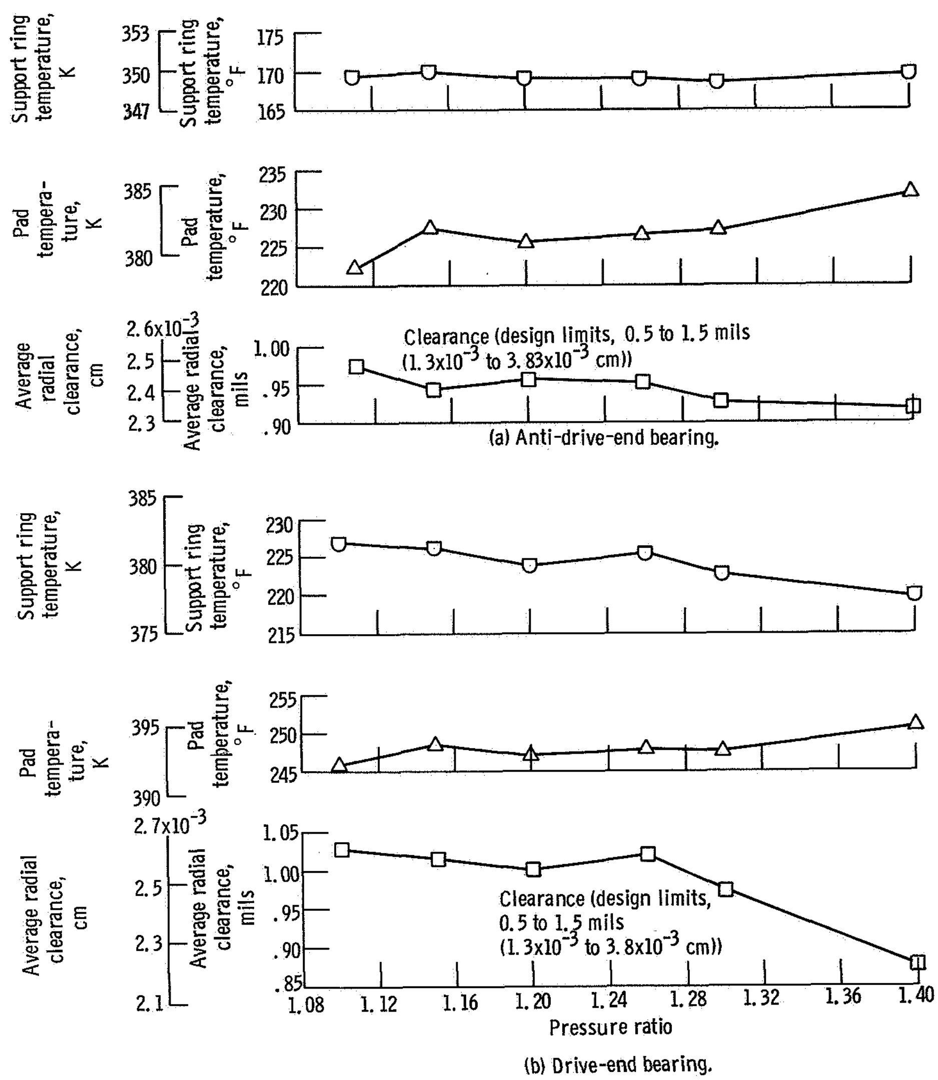


Figure 12. - Bearing parameters as function of pressure ratio. Inlet pressure, 8.45 psia (5.83×10^4 N/cm^2).

Figures 12(a) and (b) show the effects of turbine pressure ratio on bearing clearance for both journal bearings. The journal clearances are very sensitive to small temperature changes. Small changes in the liquid coolant temperatures were difficult to control precisely enough to prevent scattering of the clearance data.

Figure 12(a) shows the effect of the turbine pressure ratio on the anti-drive-end journal bearing clearance. Also plotted are the average bearing pad temperature and the average bearing support ring temperature. The pad temperature is indicative of the shaft temperature. The change in clearance with increasing pressure ratio is small, but the trend to a decreasing clearance is shown. The only temperature change causing the reduction in clearance is the increase in shaft (pad) temperature caused by increased alternator output power.

Figure 12(b) shows the effect of pressure ratio on clearance in the drive-end bearing. The overall clearance change is three times as great as that of the anti-drive-end bearing. The larger clearance change is caused by changes in the support ring temperatures as well as the shaft temperatures, the support temperature change being dominant. The support temperatures are affected by the proximity of the bearing to the turbine. As pressure ratio is increased, the turbine exhaust temperature is reduced, causing a temperature reduction in the drive-end bearing support ring.

Figure 13 shows how the bearing clearance changes as the turbine inlet is brought up to design temperature. Throughout the startup, the turbine pressure ratio and turbine inlet pressure were held constant at design conditions. The liquid coolant temperatures and flows were also maintained constant. The rate of increase in temperature was solely

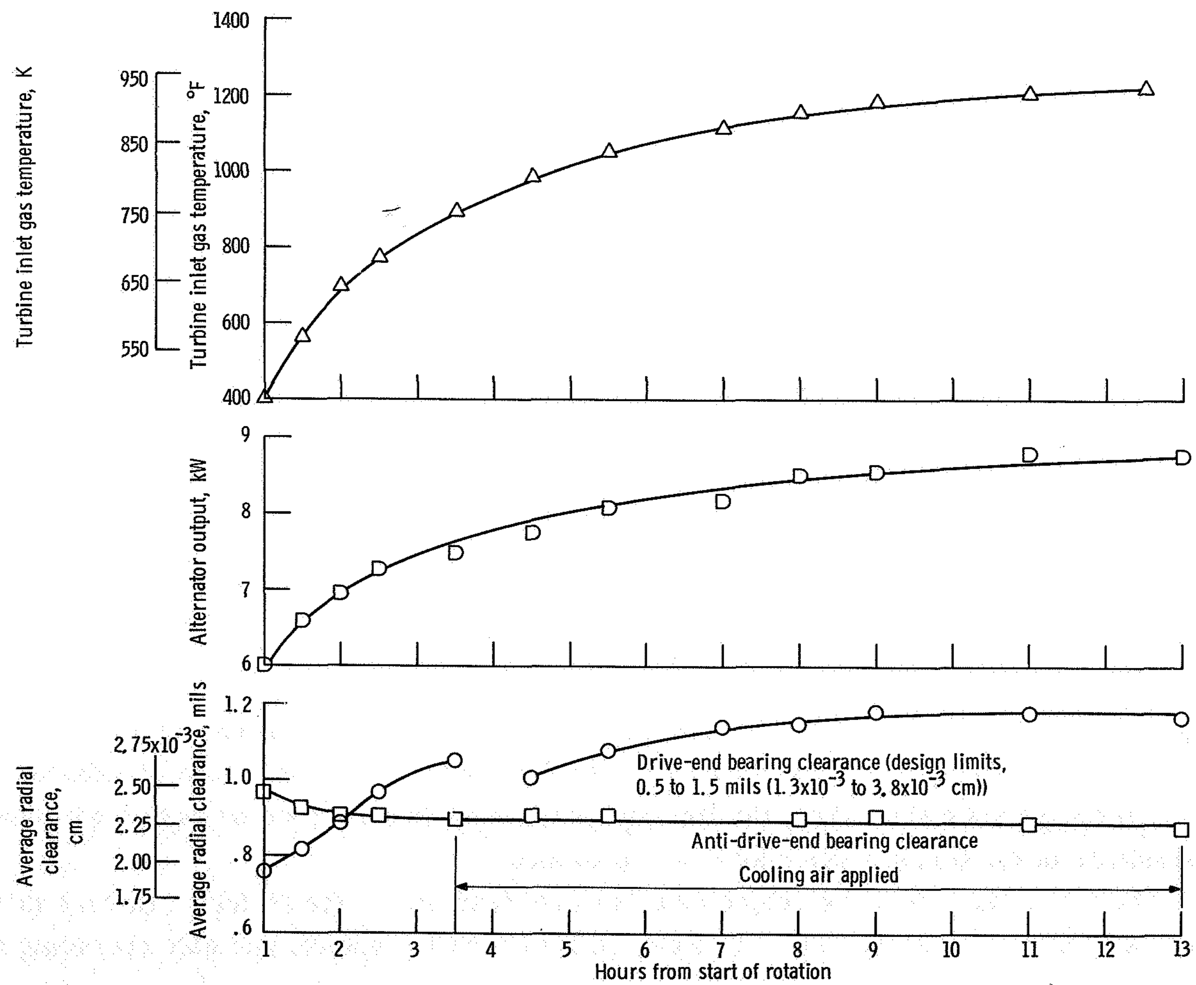


Figure 13. - Normal startup with fixed pressure ratio of 1.26 and inlet pressure of 8.45 psia (5.83x10^4 N/cm^2).

dictated by the test facility.

Note the large dependency of the drive-end bearing clearance on the turbine inlet temperature. As mentioned before, the expanding bearing mounting is the reason for the increasing clearance. The clearance in both journal bearings remained within the design operating limits throughout the startup without external adjustment.

At the $3\frac{1}{2}$-hour point during the startup, application of the cooling air to the adapter housing was required. The drive-end bearing clearance is reduced due to the cooling of the bearing mounting.

Figures 14(a) and (b) show the oscilloscope traces produced by the output of capacitance probe units. The photographs were taken when the turboalternator was operating at design conditions.

Figure 14(a) shows the orbital movement of the shaft center at both journal bearings. The orbit diameter is approximately 0.1 mil (2.5×10^{-4} cm), and the conical motion is small; thus, good shaft operating stability was obtained.

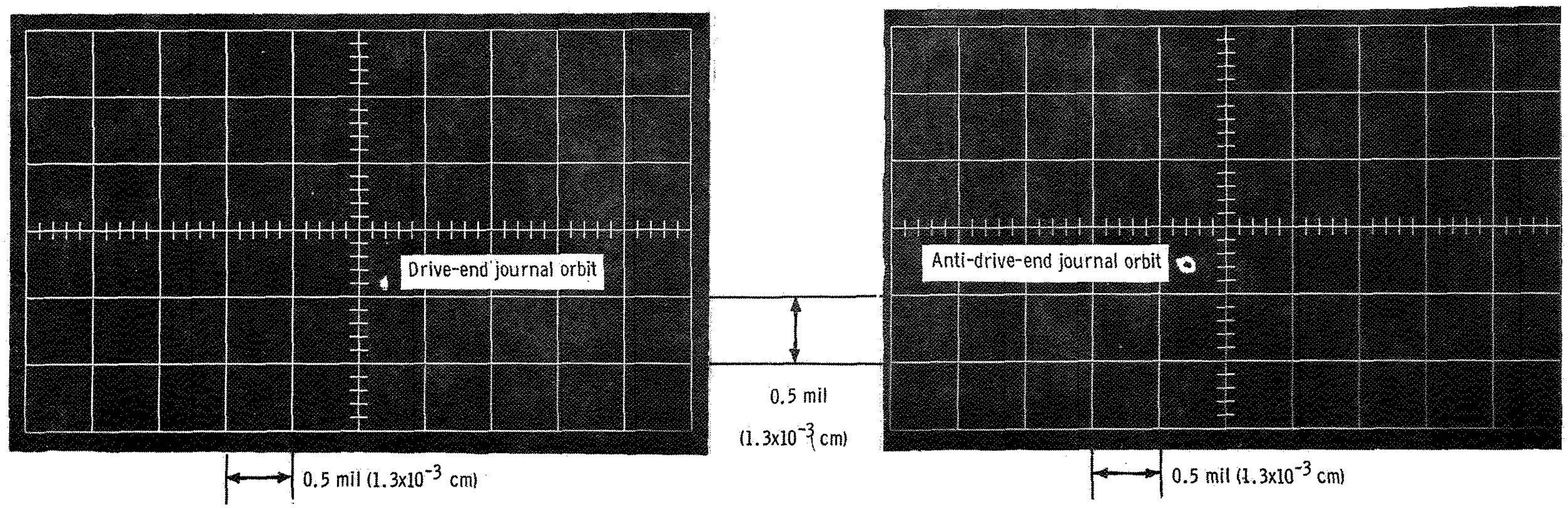
Figure 14(b) shows the bearing radial clearance on two bearing pads. The capacitance probes are mounted in the two pads opposite each other in the bearing. The "touch line" designated in figure 14(b) is the point at which the bearing pad contacts the shaft. The average radial clearance indicated by the traces in the photograph is slightly over 0.75 mil (1.9×10^{-3} cm).

The thrust bearing was operating hydrodynamically for all of the performance tests. Hydrostatic jacking gas was applied only when starting and stopping the shaft.

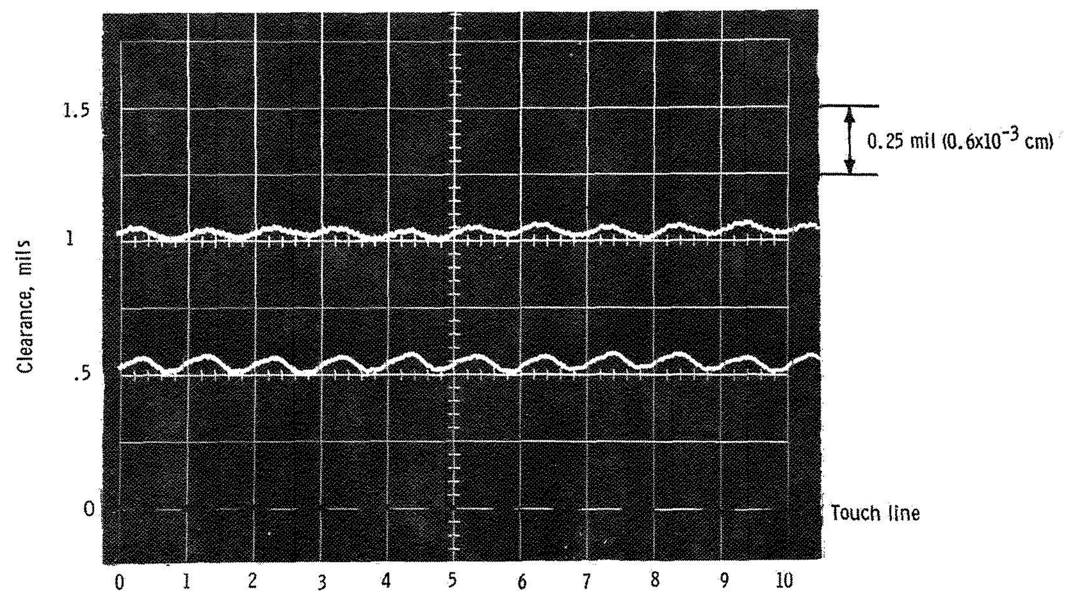
The main thrust clearance is the distance between the main thrust stator and thrust runner. The reverse thrust clearance is the distance between the reverse thrust stator and the thrust runner. The effect of turbine pressure ratio on thrust bearing clearance is shown in figure 15. The thrust load on the main thrust stator increases with increasing pressure ratio or turbine power. The absolute movement of the shaft is indicated by the reverse thrust clearance. The main thrust clearance is reduced as the load is increased but the bulk of the shaft movement is absorbed by the main thrust stator flexible mount. During the high power tests, the thrust load rose proportionately to alternator power. The main thrust clearance at 33.2 kilowatts electric was 1.1 mils (2.8×10^{-3} cm).

Figure 16 shows the main and reserve thrust clearance traces. The trace of the reverse thrust clearance shows the absolute movement of the thrust runner since the reverse thrust stator is solidly mounted. The flatness of the main thrust clearance trace in comparison shows that the flexibly mounted main thrust stator is following the movements of the thrust runner as it was designed.

The turboalternator was subjected to several transient tests at design turbine inlet pressure and temperature in order to determine overall response and safe operating re-



(a) Showing orbital movement of shaft center.



(b) Bearing radial clearance.

Figure 14 - Oscilloscope traces produced by output of capitance probes.

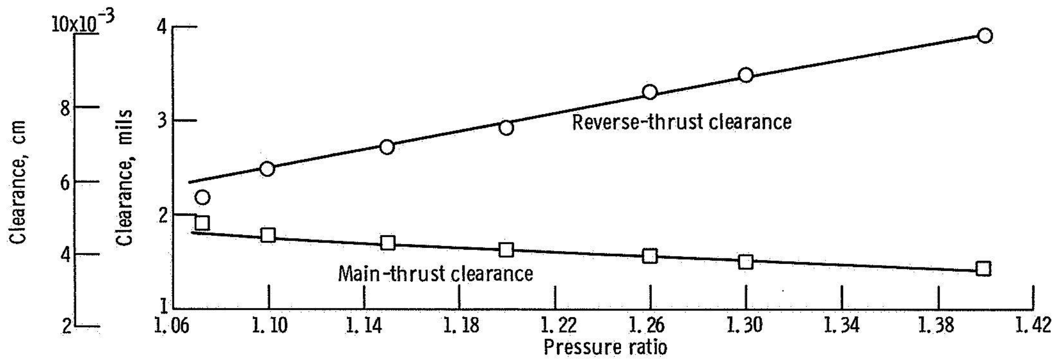


Figure 15. - Thrust clearances as function of pressure ratio.

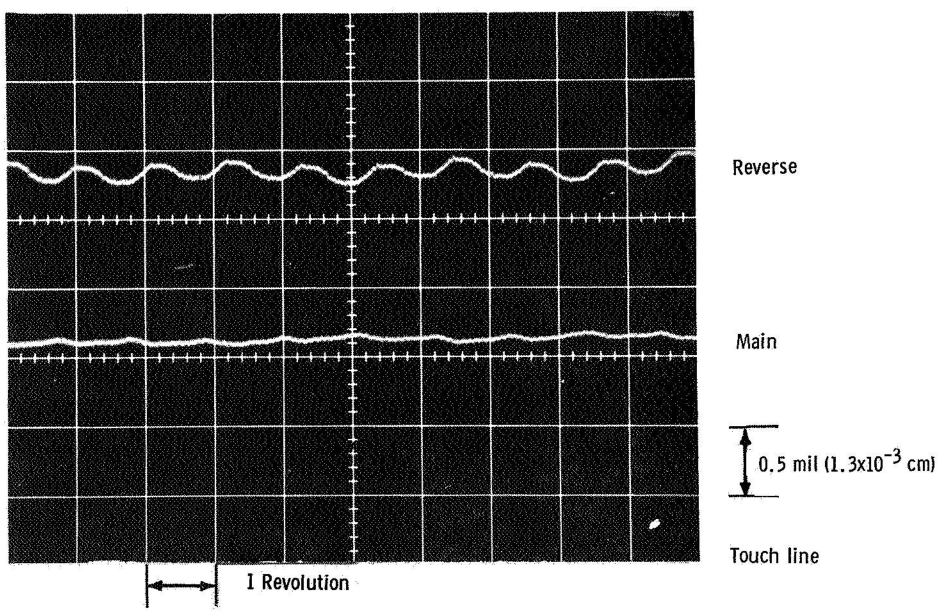


Figure 16. - Reverse- and main-thrust clearance traces.

gions for the bearings. The transients were applied by removing and adding alternator load in steps. Shaft speed was controlled by the turbine outlet valve which varied the turbine pressure ratio. During these tests, the journal and thrust bearings were operating hydrodynamically. The transients resulted in negligible journal bearing clearance changes and large changes in thrust bearing clearances. The thrust clearance changes were caused by the differential pressure change across the turbine. The largest change occurred with a phase-balanced step load of 12 kilowatts electric with an 0.8 power factor. The main and reverse thrust bearing clearances are plotted against time for the transient in figure 17.

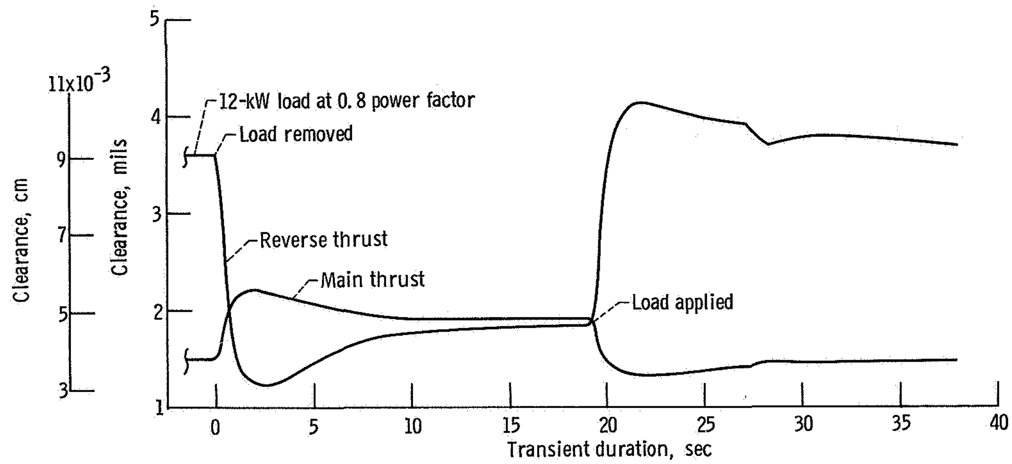


Figure 17. - Thrust bearing transient performance.

The test results indicate that the bearings operate satisfactorily with electrical transients representing an alternator design output of 12 kilowatts electric which is 3.2 kilowatts electric over the 8.8 kilowatts electric output obtained at turbine design condititions.

Periodically during the running of the turboalternator tests, turbine operating conditions were reset to the design values in order to display performance. The only changes noticed were due to instrumentation drift. Ideally, bearing deterioration, especially wear, can only be found by examining the bearing parts. But the turboalternator was not disassembled for inspection.

SUMMARY OF RESULTS

A turboalternator having a turbine inlet temperature of 1225° F (936 K) and gas-lubricated bearings was tested for 1165 hours. During this time, it was started and stopped 24 times. The tests yielded the following results:

1. The gas bearings operated satisfactorily and within the design clearance range for all design full-load electrical transients and steady-state tests. The drive-end journal bearing was affected by output power much more than the anti-drive-end bearing.

2. The alternator output at design turbine conditions was 8.8 kilowatts electric.

3. Internal temperatures at design conditions were lower than predicted.

4. The maximum output power reached was 33.2 kilowatts electric. The alternator magnetic field had not yet reached its saturation point and the highest internal alternator temperature reached exceeded the design specification by only 6° C. At the design specification temperature, the output is 32 kilowatts electric.

5. The thrust bearing operated successfully at an overloaded condition proportionate to the turboalternator output of 33.2 kilowatts electric.

6. The thrust bearing was able to absorb full-thrust load changes caused by the change in turbine pressure ratio during the full-load electrical transients.

Lewis Research Center,
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